Design, Optimization and CFD Simulation of Improved Biogas Burner for 'Injera' Baking in Ethiopia

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Abstract— **Urban and rural poor peoples in Ethiopia suffer different disease due to indoor air pollution as they are still using traditional biomass based 'injera' or bread backing activities. Over 90% of energy consumed in household level is for cooking and from this injera baking accounts for over 50%. Different scholars have been attempted to conserve the challenges of indoor air pollution by designing biogas injera baking burner nowadays. But, the external manifold diameter of currently available biogas injera baking burner was 17cm and thus due to its inefficient size, the uniformity of heat distribution throughout the baking pan is not well designed and optimized till today. Thus, this study focuses on design and optimization of improved biogas injera baking burner for uniform heat distribution of the burner throughout the backing pan. Design and optimization of the burner was employed through analytical and computational fluid dynamics (CFD) simulations of the burner by varying number and size of flame ports to increase the diameter of manifold prior to actual fabrication. To get optimum cooking heat distribution of injera baking burner optimum design size, number of holes on burner, proper mixing of air and fuel flow rate was analyzed in this study. From the result obtained it is concluded and proposed that the optimum burner manifold diameter is 26cm. Therefore, it is believed that, the study will pave the actual fabrication of the improved biogas injera baking stove and attempts to improve the life style of energy poor peoples in urban and rural areas of Ethiopia through reduction of traditional inefficient biomass burning.**

Keywords*— *Biogas burner, Injera baking, optimization, CFD simulation

I. INTRODUCTION

Injera is the cultural staple bread food item in Ethiopia and made from indigenous grain called "Teff". Baking this food item in the traditional three stone stoves consumes huge amount of firewood in local areas and this leads different localities to continuous deforestation of trees, exposing of child and women's to smoke and exposing the environment to global warming due to its inefficiency. Accordingly, over 90% of energy consumed in household level accounts for cooking and from this injera baking accounts for over 50% **[1]**, **[2]**. The systematic depletion of forest coverage has incurred in the scarcity of fuel wood and has menaced soil fertility and aquifers protection. This situation becomes critical for the rural households since they rely almost entirely on firewood and base most of their income on their agricultural yield **[3]**.

To mitigate the above challenges, different scholars have been attempted to design and develop injera baking biogas stoves. But, due to its inefficient size and non-uniformity of heat distribution throughout the baking pan of the currently available 17cm burner manifold diameter it needs better optimization. Hence this project aimed to design, optimization and CFD simulation of injera backing biogas burner to enhance its performance. The gap observed in the previous designs was; heat transfer through the stove, heat loss and high fuel consumption are the major problems **[4]**. According to the study of **[5]** comparison between metal Mitad and traditional clay Mitad (local name of injera baking plate made from clay soil) with biogas stove a slight difference formed on the eyes (holes) on the pancake's surface. This is due to unequal distribution of heat throughout the baking pan. Another researcher also reported in his work on optimization and construction of baking pan and biogas stove; because of small burner's manifold diameter the heat distribution through baking pan is not uniform without manual stove adjustor **[6]**

The rural household energy problem in many developing countries like Ethiopia has not changed for many years, and millions of people still lack enough energy demand to sustain economic development. Due to this fact, this project has been designed and on the way to be implemented in Ethiopia after intensive experimental testing of the burner. Currently, this work is only scoped to present the design and optimization of the device thermal distribution using analytical analysis and CFD simulation. Accordingly, the analytical design and CFD simulation is presented as follows in section II and section IV respectively.

Methodology involved in this study is analytical design analysis, geometrical design using Solidwork Workbench design tool and CFD analysis using ANSYS fluent version 14. The CFD was preferred because of its flexibility, very cheapness and easiness to accomplish geometrical design and flow visualization prior to actual fabrication. But, as the accuracy of computational results need validation against analytical or experimental results, the work is compared and validated with analytical results of this study. The experimental performance testing is remained as the further work (next work) of this study.

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II. ANALYTICAL DESIGN AND OPTIMIZATION

The design analysis of biogas injera backing burner involves the determination of the following important parameters like: injector orifice (jet) size, throat size, burner port, diameter of the jet ($\boldsymbol{d_j}$), length of the air intake holes measured from the end of the jet, length of the mixing pipe, number and diameter of flame port holes, and height of the burner head.

The analytical design procedure for the biogas burner was adopted from Shewangizaw W. et al (2016) **[7]** and employed in the design of injera baking biogas burner. Accordingly, the total output power required ($\mathbf{P_{out}}$) for injera baking is around 1.5kw. Currently, the efficiency of big sized biogas burner in Ethiopia is 51% with biogas pressure of 8mbar and 45.1% with biogas pressure of 7.47 mbar **[8]** Here the average efficiency ($\boldsymbol{\eta_{av}}$) has taken to be 48.05%. Thus it is possible to calculate the power required and biogas volumetric flow rate for injera baking application based on output power and average stove efficiency as follows:

$$P_{requiered} = \frac{\mathrm{P_{out}}}{\eta_{\mathrm{av}}} \quad (1)$$

Using "(1)" the power required for injera baking application should be 3.12kw. Energy require for injera baking for one hour should be: $3.6 \times 3.12\mathrm{MJh^{-1}} = 11.25\mathrm{MJh^{-1}}$.

It is reported that, Calorific Value of Biogas (CV_{Biogas}) is $22MJ/m^3$ [8] based on required energy value and calorific value it is possible to calculate biogas flow rate (Q_{Biogas}) as follows:

$$Q_{Biogas} = \frac{\mathrm{P_{required}}}{CV_{Biogas}} \quad (2)$$

Hence, using "(2)" the biogas flow rate is $0.51\mathrm{m^3h^{-1}}$.

A. Determinatio of injector orifice (jet) size

The injector orifice area was calculated from empirical version of Bernoulli's theorem which defines the gas flow rate through an injector orifice (jet) as follows:

$$Q = 0.04601 C_D A_j \sqrt{\frac{p}{s}} \quad (3)$$

Where: Q (gas flow rate) $=0.51\mathrm{m^3h^{-1}}$

$\boldsymbol{A_j}$= area of orifice ($\mathrm{mm^2}$)

P= (gas pressure before orifice) =8 mbar

s = specific gravity of gas = 0.94

$\boldsymbol{C_D}$= (coefficient of discharge for the orifice) = 0.94

Assuming suitable injector with a C_D of 0.94 including the vena contractor and friction losses through the orifice and a gas supply pressure of 8 mbar, the injector orifice size or diameter of the jet (d_j) is 2.3 mm as shown below using "(3)".

$$. \; d_j = \sqrt{\frac{Q}{0.0361 \times C_D}} \times \sqrt[4]{\frac{s}{p}} = 2.3mm \quad (4)$$

Then using "(4)" the area of orifice jet $\left(\boldsymbol{A_j} = \frac{\boldsymbol{\pi d_j^2}}{\mathbf{4}}\right)$ is $4.057\mathrm{mm^2}$.

Finally, using $\boldsymbol{A_j} = 4.057\mathrm{mm^2}$, the velocity of the biogas in the orifice jet $\left(\boldsymbol{v_j} = \frac{\boldsymbol{Q_{Biogas}}}{\mathbf{3.6 \times 10^{-3}} \boldsymbol{A_j}}\right)$ is $35.01\mathrm{ms^{-1}}$.

The following graph ("Fig. 1") shows graph of biogas flow rate versus pressure at orifice using "(3)".

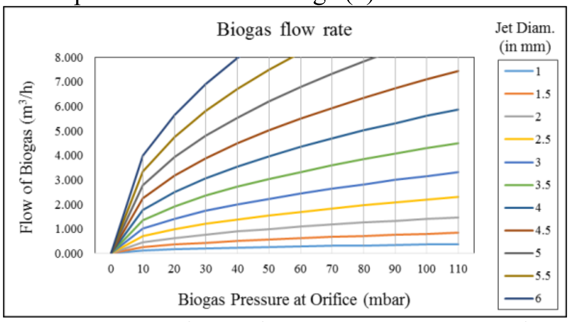

Fig. 1. The graph of biogas flow rate versus pressure at orifice

B. Determination of throat size

From the composition of biogas, the stoichiometric air requirement is 5.7, and then the entrainment ratio (r) should be half of air requirement, that is equal to 2.85.

The flow rate of the biogas and air mixture at optimum aeration is given by:

$$Q_m = Q_{Biogas}(1 + r) = 5.45 \times 10^{-4} m^3 s^{-1} \quad (5)$$

Throat diameter (d_t) is calculated using Prig's formula including orifice diameter as follows:

$$d_t = \left(\frac{r}{\sqrt{s}} + 1\right) d_j = 8.95mm \quad (6)$$

However, it is better to use the stoichiometric value of primary air of 5.7 directly rather than using r = 2.85 to get better aeration and control using primary air flow adjuster. So, the better design diameter of throat will be 15.63mm. Then the throat area becomes $191.98\mathrm{mm^2}$.

The air inlet ports must have an area similar to that of the throat [9].

The gas pressure in the throat ($\boldsymbol{P_t}$) can be calculated as:

$$P_t = P_j - \rho_{Biogas} \frac{v_j^2}{2g}\left[1 - \left(\frac{d_j}{d_t}\right)^4\right] = \left(10^5 - 75\right)pa \quad (7)$$

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The value of P_j is 10^5 pa as the throat is open to the air. This pressure drop is sufficient to draw primary air by the air inlet parts to mix with the gas in the mixing tube [9].

Calculating the Reynolds number should be used to check the pressure drop due to the flow of biogas and air mixture in the mixing tube:

$$\mathrm{Re} = \frac{\rho_m d_t v_t}{\mu} = \frac{4\rho_m Q_m}{\pi \mu_m d_t} = 3111.12 \tag{8}$$

Where ρ_m and μ_m are the density and viscosity of the mixture ($\rho_m = 1.15 kgm^{-2}$& $\mu_m = 1.71 \times 10^{-5} pa\ at\ 30℃$) respectively.

Re>2000 (flow of biogas and air mixture in the mixing tube is turbulent), so the friction loss is calculated as $\boldsymbol{f = \frac{0.316}{Re^{1/4}} = 0.04231}$ and pressure drop ($\Delta \boldsymbol{P}$) could be calculated as:

$$\Delta \mathrm{P} = \frac{f}{2}\rho_m v_t^2 \frac{l_m}{d_t} = \frac{f}{2}\rho_m \frac{16Q_m^2}{\pi^2 d_t^5} l_m = 2.8 pa \tag{9}$$

This pressure drop is much lower than the driving pressure or the pressure in the throat.

C. *Determination of burner port design*

Biogas has a stoichiometric flame speed of only 0.25 m/s Burning velocity. The mixture supply velocity ($\boldsymbol{v_p}$) is given by **[9]**:

$$v_p = \frac{Q_m}{A_p} << 0.25 m^{-1} \tag{10}$$

Where: A_p (the total burner port area in m^2) = $n_p \frac{\pi d_p^2}{4}$

$\boldsymbol{n_p}$: Number of ports
$\boldsymbol{d_p}$: Diameter of each port in m

The total burner port area will be chosen as:

$A_p > \frac{Q_m}{v_p} > 0.00218 m^2$, assume 30% area was added;

$$A_p = 0.00218 \times (1 + 30\%) \times 10^6 mm^2 = 2834 mm^2$$

Fulford (1996) **[9]** and Itodo (2007) **[10]** used 5mm and 2.5mm diameter holes respectively. However, a problem of flame lift was recorded **[11]**.

Using 2mm port diameter to minimize the problem of flame lift, the total number of required ports will be:

$$n_p > \frac{4A_p}{\pi_p^2} = 902 \tag{11}$$

Using flame stabilization, it should be possible to reduce this number of burner ports, by up to 1/5 **[9]**, so ~ 180 holes will be sufficient.

Among these 180 holes 137 holes can be used for the outer manifold, with hole diameter of 2mm and 4 mm gaps between holes, arranged in a circular pattern, gives a total outer circumference of 137 × (2 + 4) = 822 mm. So that the holes centers will be placed around a circle of the outer diameter (D=822 mm/π) =261.7mm (~26 cm).

Then the remaining 43 holes can be used for the inner manifold diameter, with hole diameter of 2mm and 4 mm gaps between holes, arranged in a circular pattern, gives a total inner circumference of 43 × (2 + 4) = 258 mm. So that the holes centers are then placed around a circle of inner diameter (d= 258 mm/π) =82.17mm (~8cm) as clearly shown in "Fig. 2" and "Fig. 5".

III. Geometry and Design Modeling of Burner Port

The burner is designed using the analytical results presented in the above section II. The total diameter of the new improved cost effective and efficient burner is 260mm as its design drawings are shown in *"Fig. 2"*.

The geometry is created using solid work software as shown in *"Fig. 3"*.

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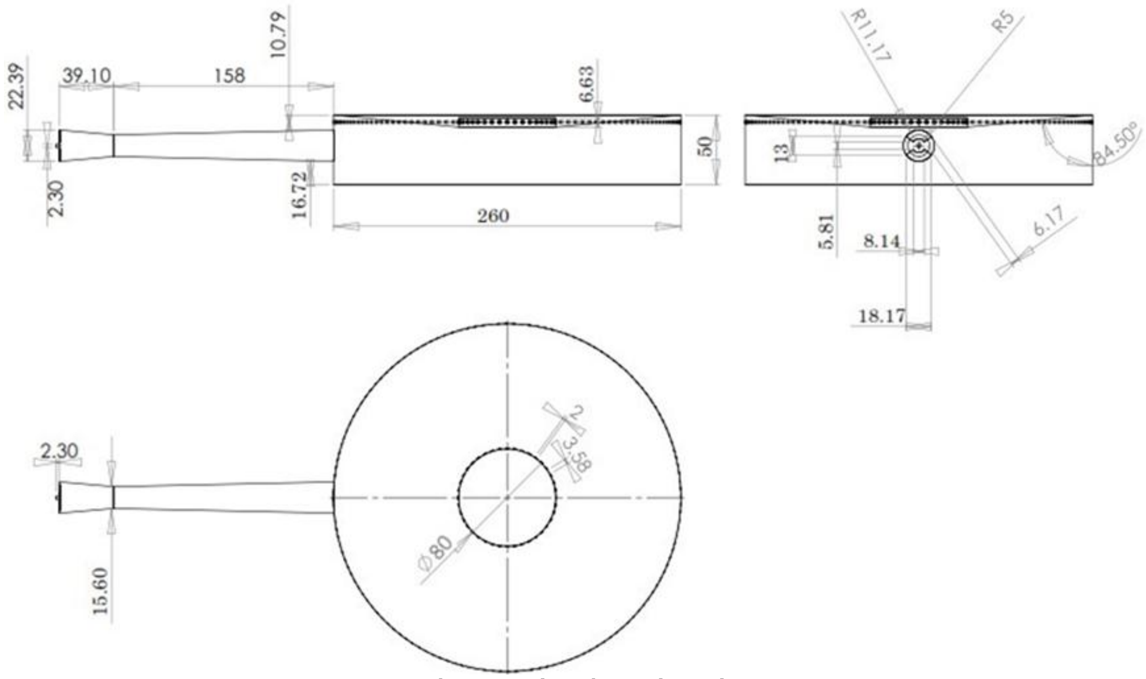

Fig. 2. The improved new burner design drawing

A. Design Comparison between existing and newly proposed burner

Using analytical values of burner discussed above the burner 3D geometry is created with solid work software provided that the material is 2mm thick sheet metal as shown in “Fig.3” below.

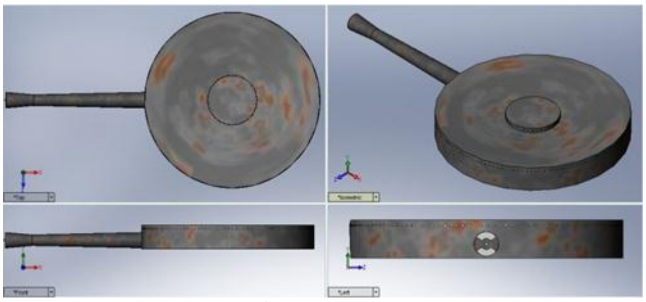

Fig. 3. Solidwork design of new burner with 2mm thick sheet metal

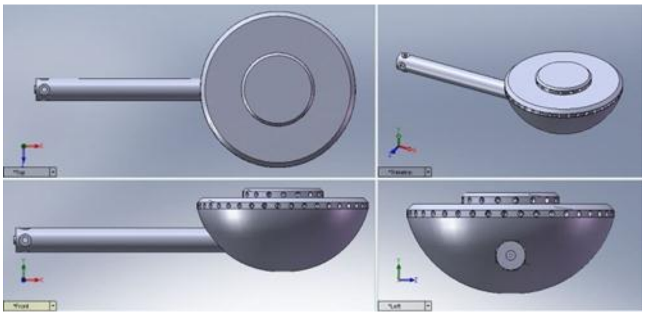

Fig. 4. Solidwork design of previous burner constructed with aluminum [12]

Existing and new design specification parameters of important part of the burners was compared in detail in Table I and their solid work design of the burners are shown in “Fig. 3” and “Fig.4”. There are some variations in certain features (e.g. number and diameter of ports, manifold diameter and used material type) between newly improved and previous burner design.

TABLE I. DESIGN COMPARISON BETWEEN NEW AND PREVIOUS BURNER

	Material		Injector Size	Primary air inlet			Gas-air mixing tube			Port			Manifold	
	Type	Thickness (mm)		No. of holes	Area of holes (mm^2)	Air inlet angle with respect to injector	Length (mm)	Throat diameter (mm)	Angle of taper into and away throat	No. of port	Diameter of port (mm)	Distance b/n port (mm)	External diameter in cm	Shape
Previous	Aluminum	5	2	4	63.62	90^0	180	18	0^0	52	5	5	17	Cup
New	Sheet metal	2	2.3	2	96	0^0	158	15.63	5^0	180	2	4	26	Cylindrical

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IV. CFD SIMULATION AND OPTIMIZATION OF THE BURNER

A. Simulation Procedures

The Computational model of the burner was started with designing of the 3D burner using solidwork workbench design software and exported to ANSYS fluent version 14, commercial CFD software using its built-in model and algorithms. The computational mesh was created in ANSYS geometry meshing and finally the optimized simulated solution was obtained using fluent solution setup [13], [14].

The steps of fluent solution setup setting for model was adapted from [15], which is the most important since these parameters setting was related to the simulations condition including energy, radiation, turbulent viscous and species. The energy equation must be "on" since the case having the temperature change in the combustion process. The turbulent viscous model selection is realizable "k-epsilon" with standard wall function as near wall treatment. The discrete ordinate (DO) is chosen for the radiation and non-premixed is for species. Probability density function (PDF) mixture is used for the material and the species count, it is depending on the model setting. Also in PDF mixture under chemistry tab energy treatment was selected non-adiabatic condition.

B. Boundary Conditions determination

The boundary conditions as shown in "Fig. 5" are used to represent the flow in the burner for the CFD analysis. After defining proper boundary conditions, the design of the gas mixer is imported to Fluent for carrying out further simulation. The species model used to calculate various species formation during combustion is non–premixed combustion model. Initial velocity conditions of biogas and air are considered as 35.01 m/s and 5.2 m/s respectively. The PDF table creation boundary condition for the fuel and oxidant was set as shown in Table II. The biogas fuel was created by the mixing of methane (60%) and carbon dioxide (40%) on molar basis. For the oxidant, the mixing of nitrogen (79%) and oxygen (21%) was used as the normal combustion air configuration. The fuel and oxidant temperature was set at 300 K. Finally, CFD simulation has been performed by varying the geometry and size of the burner manifold to get optimum size to validate the analytical result discussed under section II.

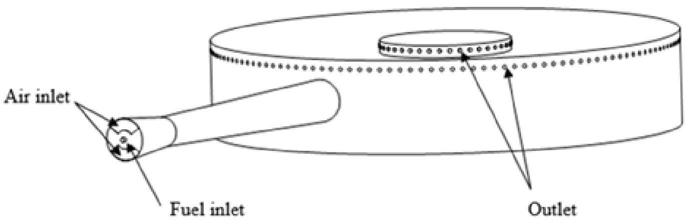

Fig. 5. Burner boundary conditions

TABLE II. PDF TABLE CREATION FOR BOUNDARY CONDITION PARAMETERS

Species	Fuel	Oxidant
CH_4	0.6	0
H_2	0	0
N_2	0	0.79
O_2	0	0.21
CO_2	0.4	0

V. RESULTS AND DISCUSSION

A. Temperature variation with position

The following "Fig. 6" represents the plane contours of temperature. From the figure we can say the temperature of mixing chamber at inlet was not greater than the room temperature which was similar to our boundary conditions. But with small position change the maximum temperature obtained is 1,775k and then slightly decreases and remains uniform throughout mixing chamber. The graph, "Fig. 7" shown below indicates, as the mixture of gas and air stored in the manifold the flame point combustion temperature increased slowly with circumferential position (x-axis) till it reached a maximum temperature of 1,775K and then slightly decreases.

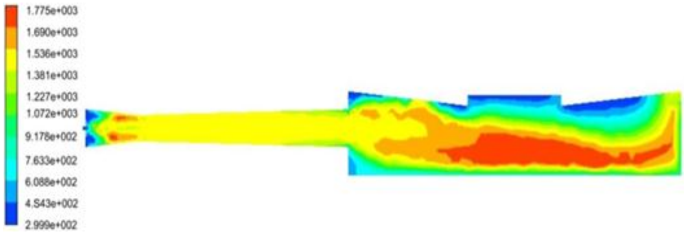

Fig. 6. Contours of temperature distribution (k)

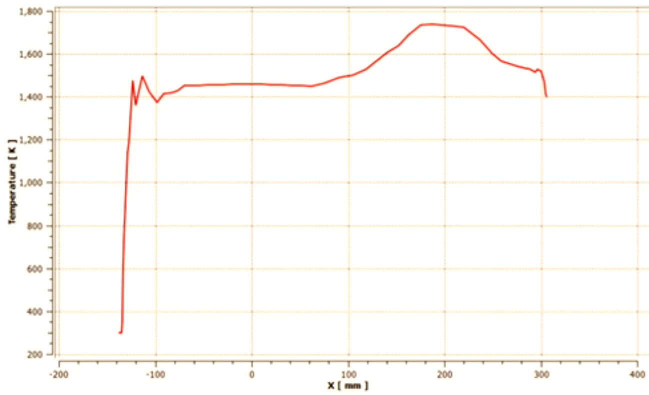

Fig. 7. Temperature distribution with position along x-axis

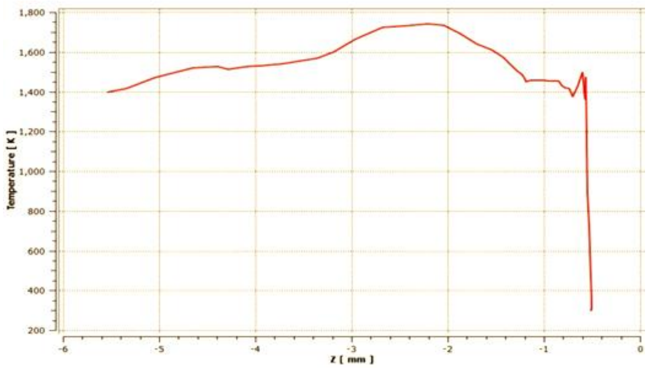

Fig. 8. Temperature distribution with position along z-axis

B. Velocity variation with position

Velocity of the mixing species is important from the point of view of fuel and air transport which comprises an integral part of combustion. As shown in "Fig. 9" and "Fig. 10" the velocity of fuel and air decreases throughout mixing chamber and remains almost uniform throughout manifold. This is in a good agreement with various literatures result and

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confirms uniformity of heat distribution throughout the baking pan. i.e., it shows the emerging of velocity from the injector enters the end of the mixing tube in a region called the "throat". The throat has a much larger diameter than the injector, so the velocity of the gas stream is much reduced as expected.

Hence, the CFD result confirms that, the optimum manifold diameter of the burner is 26cm to distribute heat uniformly throughout the baking pan. As per this manifold diameter 5cm thickness with 60cm diameter of insulation and 54cm diameter with 13mm thickness of baking pan is obtained to be optimum which is almost equal with the size of hotel injera 50cm in size.

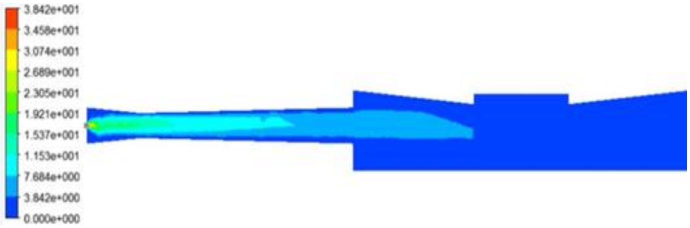

Fig. 9. Contours of velocity (m/s)

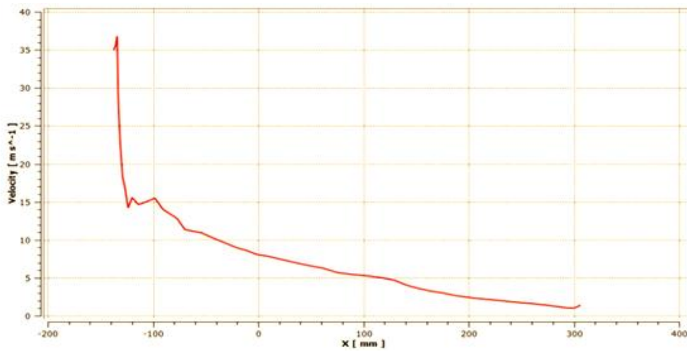

Fig. 10 Velocity variation with position

The post processing of the CFD result in (Fig. 11 and Fig. 12) shows the 3D velocity streamline for all domains with 100 point sampling equally spaced start from air inlets with forward and backward direction as in "Fig.11". And "Fig. 12" shows with the same setting, but it start from fuel inlet. The streamline of air inlet is more than the fuel inlet since the volume of air inlet is higher than the fuel inlet as expected.

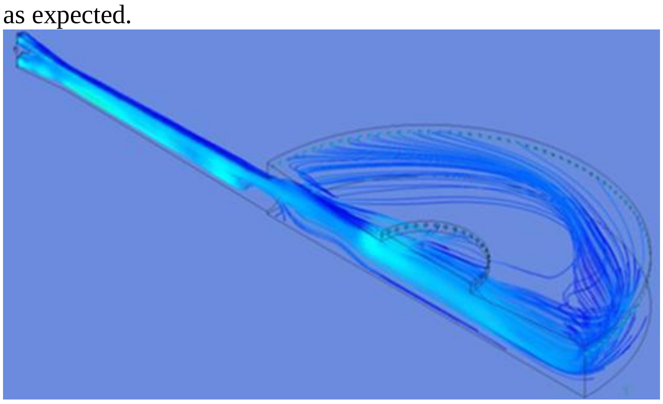

Fig. 11. Velocity streamlines start from air inlets

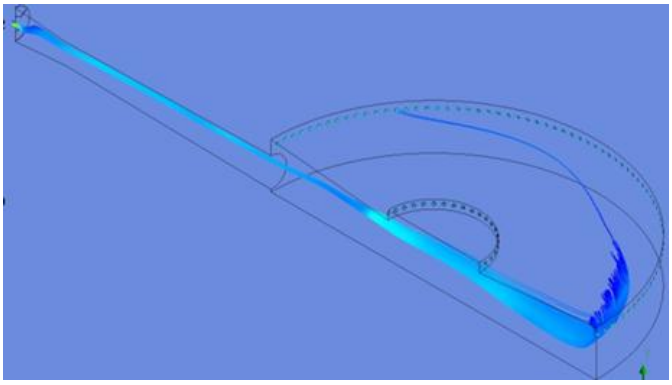

Fig. 12. Velocity streamline start from fuel inlet

VI. CONCLUSIONS

The article demonstrates one of the possibilities of using CFD in terms of finding optimum burner manifold size and arrangement of holes on burner manifold. It mainly focused on the analytical design and optimization of biogas burner with CFD simulation. To this end, from the study result it is observed and proposed that, the optimum manifold diameter is 26cm to distribute heat uniformly throughout the baking pan. As per this manifold diameter 5cm thickness with 60cm diameter of insulation and 54cm diameter with 13mm thickness of baking pan is obtained to be optimum which is almost equal with the size of the local standard hotel injera 50cm in size.

Therefore, it is believed that, the study will pave the actual fabrication and implementation of the improved biogas injera baking stove and attempts to improve the life style of energy poor peoples in urban and rural areas of Ethiopia through reduction of traditional inefficient biomass burning. Furthermore, the results of this study will provide valuable data for biogas injera baking stove manufacturers.

Further work will be:

- Fabrication and performance testing of improved biogas burner which integrated with the total parts of injera baking stove (like: baking pan ('Mitad'), support, cover insulation and stand), and
- Economic viability analysis of this injera baking stove in Etiopia.

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